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# CHARACTERIZATION OF STIFFENING AND DAMPING MATERIALS FOR PLANING BOAT SHOCK MITIGATION

R. S. PETERSON
COASTAL RESEARCH AND TECHNOLOGY DEPARTMENT

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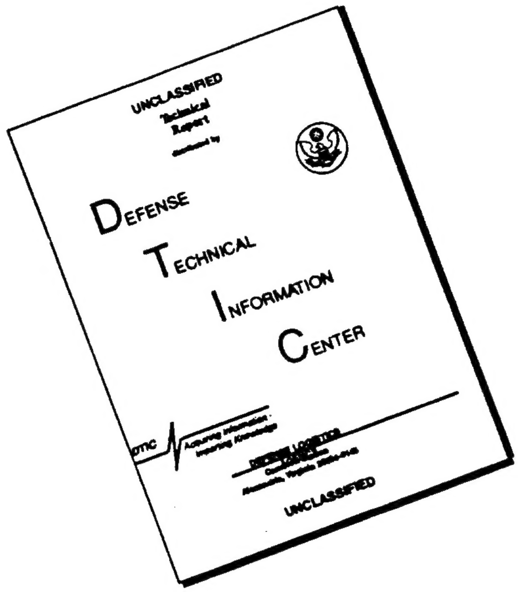


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R. S. Peterson

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Attn: Code 120
Commanding Officer
CSSDD NSWC
6703 W. Highway 98
Panama City, FL 32407-7001

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The occupants of high-speed planing boats operating in heavy seas are subject to repeated shock loads that can injure or impair performance. Identification and evaluation of shock mitigation concepts requires development of a nonlinear, time-domain, wave-to-wave elastic planing boat/human dynamic simulation. Shock mitigation concepts include the use of various stiffening and damping materials in the cockpit or within the hull structure. This report summarizes a general method for characterizing stiffening and damping materials and devices for use in a time-domain planing boat simulation, and derives the equations for the stiffening characteristics of several materials and devices.

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## FOREWORD

The occupants of high-speed planing boats in heavy seas are subject to repeated shock loads that can injure or impair performance. Identification and evaluation of shock mitigation concepts requires development of a nonlinear, time-domain, wave-to-wave elastic planing boat/human dynamic simulation. Shock mitigation concepts include the use of various stiffening and damping materials in the cockpit or within the hull structure. This report summarizes a general method for characterizing stiffening and damping materials and devices for use in a time-domain planing boat simulation, and derives the equations for the stiffening characteristics of several materials and devices.

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This report has been reviewed by D. C. Summey, Head, Hydrodynamics and Navigational Guidance and Control Division.

Approved by:

GARY J. KEKELIS, Head
Coastal Research and Technology Department

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## INTRODUCTION

Since FY93 the Coastal Systems Station (CSS) has been developing technology to reduce shock loads experienced by operators of high-speed planing boats in support of the Office of Naval Research. Methods under consideration include isolation materials and devices used in the cockpit environment, isolation materials and devices between the cockpit and the lower hull, and hull shape. This report, which addresses the first and second categories, summarizes a simple method for characterizing the isolation characteristics of stiffening and damping materials and devices and derives the stress-strain equations for several stiffening devices.

## MODELING ISSUES

During the course of the shock mitigation investigation, CSS developed a time-domain hull impact code called Water-Entry Dynamics and Injury Model (WEDIM). WEDIM simulates the vertical water-impact, submergence, and emergence of a two-dimensional, prismatic, elastic hull segment with seated humans. As shown in Figure 1, the model is a 3 degree-of-freedom (DOF) simulation, in that the vertical dynamics of three masses is considered. (Note that all figures and tables can be found at the end of the text.) The upper mass is the seated human. The middle mass is the upper boat, or deck, on which the human is seated. The lower mass is the lower hull, which impacts the water. Each of the three masses is assumed rigid.

The stiffening and damping characteristics between the deck and the human torso center of mass are assumed to be linear. The values of the stiffening and damping constants are based on historical aircraft seat ejection investigations.[1,2] In this simple model, the shock isolation components that may be readily varied are the stiffening and damping material between the deck and the lower hull.

Two approaches for modeling the stiffening characteristics of the isolation component within a time-domain simulation such as WEDIM are to use: (1) a spring constant, $K$ (the coefficient in the linear displacement-versus-force term) or, (2) stress-strain characteristics of the component, if the material and loads are expected to produce nonlinear behavior.

WEDIM models the damping characteristics in a linear manner, requiring the damping constant (the coefficient in the linear velocity-versus-force term). The approximate method for obtaining this constant in the absence of experimental stress-strain-rate data will be described in a later section.

## STIFFENING CHARACTERISTICS

The shock isolation materials and devices that have been seriously considered by CSS include foams, helical springs, air bladders, and one-way and two-way air springs. In general, all of these devices display nonlinear and asymmetric stress-strain characteristics. The possible exception is the spring, which, if properly designed, selected, and installed, is essentially linear and symmetric about zero strain.

The stress-strain (stiffening) characteristics of a material or stiffening device, including nonlinearities, are readily measured from a sample of the material and are often provided by the manufacturer. Alternatively in the case of conventional metal springs and air springs, the spring constant and stress-strain characteristics may be estimated from geometry, air pressure, and material properties.

The equations to predict the spring constant and stress during compression and extension of helical springs are presented in a later section. A properly selected helical spring behaves nearly linearly within its design range. A spring in extension will remain essentially linear until it reaches its elastic limit beyond which it has been overstressed and may retain a set. A properly designed spring in compression will reach its *solid height* before the material yield stress is reached.

For stiffening materials such as foams, the basic material property required to estimate the linear spring constant is Young's modulus, $E$. The spring constant is then computed from the cross-sectional area $A$ and the thickness $T$ as

$$K = E\ (A/T). \tag{1}$$

Young's modulus is the material property independent of geometry, and the spring constant represents the linear stiffening characteristics of the material with specified geometry.

The stress-strain characteristics of a commercially available foam product called *Sorbothane* are shown in Figure 2 as an example. The slope of the curve at zero strain is Young's modulus. Sorbothane, often used as floor mats in factories to absorb uncomfortable and fatiguing vibration transmitted from the floor to the feet, is known for its high damping characteristics.

The stress-strain characteristics of air springs and air bladders may be highly nonlinear and, in the case of an air bladder or single-acting air spring, highly asymmetric about zero strain. The equations describing the nonlinear stress-strain characteristics of air bladders and single- and double-acting air springs are presented in a later section.

## DAMPING CHARACTERISTICS

The stress-strain-rate (damping) characteristics of a material are more difficult to measure than the stiffening characteristics. With an ideal viscous damper, the force is related to the velocity of the strain, often the result of fluid flow through an orifice. The force-versus-velocity relationship may in certain cases be modeled; however this is beyond the scope of the present effort.

Of the various materials and devices considered in this study, only certain of the foams possess significant damping characteristics. A simple approach sometimes adopted by the foam manufacturers is to characterize the damping characteristics by means of the damping ratio $\zeta$. The damping ratio is treated as a basic material property independent of thickness and area, analogous to the Young's modulus for the stress-strain characteristics. In reality, the damping depends also on displacement frequency and amplitude; however, assuming a constant damping ratio for the material is often acceptable.

Manufacturers measure the rate of decay of free vibrations and relate the rate of decay to the damping ratio. The damping ratio $\zeta$ is the ratio of the actual damping coefficient $C$ and the critical damping coefficient $C_c$, the value of the damping coefficient just sufficient to eliminate oscillatory behavior of the system. The critical damping coefficient may be easily estimated, assuming a single DOF spring/mass/damper system, where the mass $M$ and linear spring constant $K$ are known. The linear damping coefficient is then computed from the damping ratio and the critical damping coefficient.

## EXAMPLE -- SORBOTHANE

Suppose the desire is to model the damping and nonlinear stiffening characteristics of a 2.0 sq ft by 0.5 ft thick piece of Sorbothane, which supports a 200-lb weight. The manufacturer of Sorbothane foam reports an experimentally determined damping ratio $\zeta$ of 0.3 and a Young's modulus $E$ of 5190 lb/sq ft. This is the slope at zero strain in Figure 2. From Equation (1), the linear spring constant $K$ is $E\ (A/T)$, or 20,700 lb/ft.

The manufacturer also reports the stress corresponding to several nonzero strain values, as shown in Figure 2. This nonlinear stress-strain data may be called by the integration routine within a time-domain simulation, in a look-up table fashion. For example, with the thickness $T$ specified, an instantaneous extension $\delta$ of 0.25 ft yields a strain $\epsilon$ of $\delta/T$, or -50 percent. From Figure 2, the corresponding material stress $\sigma$ is 2016 lb/ft$^2$. Finally, the stiffening force $F$ is the product of the stress and the area, or 4032 lb.

The damping characteristics, as described earlier, are modeled in a linear manner. From the linear equation describing the free vibration of a single DOF spring-mass-damper, the natural frequency $\omega_n$ of the 200 lb weight on the Sorbothane is $(K/m)^{1/2}$, or 58 rad/sec. Next, again based on the linear single DOF equation, the critical damping $C_c$ is $2m\omega_n$, or 717 (lb/ft/sec). The damping coefficient $C$ is then computed as $C = \zeta\ C_c$, or 207 lb/(ft/sec). The damping force is the product of

the damping coefficient and the instantaneous strain rate computed during the time-domain simulation. The damping characteristics remain linear, based on the linear model of the weight and foam.

## CHARACTERISTICS OF SPECIFIC MATERIALS AND DEVICES

The following are primarily examples of stiffening materials. With the exception of certain of the foams, these materials and devices are lightly damped. Additional damping, if required, may necessitate the use of a rate-dependent force producer such as a "dashpot."

### FOAMS

Most foams are lightly damped and are primarily used to provide stiffening. Examples of foams that have been used or considered for planing boat shock mitigation include Airex, Fabcel Pads, Ensolite, CONFOR, ISOLOSS LS, and Sorbothane. As mentioned earlier, the manufacturer of Sorbothane claims one of the highest damping ratios of all isolation foams.

The stiffening properties of a given foam are not readily predicted. Thus, the usual approach in modeling foam is to measure the static force required to deflect a sample of the material with a given area and thickness, over a range of several deflections, and convert the values to stress and strain. Some of the manufacturers provide stress data over a range of strain values. The damping ratio of the foams ranges from about 0.1 to 0.3.

### HELICAL SPRINGS

The linear stiffening characteristics of a round-wire helical spring may be estimated analytically.[3] The user specifies the material torsional modulus $G$ and ultimate strength $S_u$, wire diameter $d_w$, coil outside diameter $D_c$, number of coils $N_c$, and spring free length $L_f$. The torsional yield strength $S_{ty}$ has been shown for helical springs to be approximately 44 percent of the ultimate strength. The spring constant is

$$K = \frac{d_w^4 \, G}{8 \, N_c \, D_{cm}^3} \qquad (2)$$

where $D_{cm}$ is the mean coil diameter, $D_c$ - $d_w$. The force required to compress the spring to a solid is

$$F_s = K\ (L_f - L_s) \tag{3}$$

where $L_s$ is the solid length estimated as

$$L_s = d_w (N_c + 1)\ . \tag{4}$$

The spring *stress-correction factor* is

$$K_{sc} = \frac{4C - 1}{4C - 4} + \frac{0.615}{C}\ , \tag{5}$$

where $C$ is the *coil curvature index* $D_{cm}/d_w$. The shear stress associated with deflection of the spring under a load $F$ is

$$\tau_s = K\ \frac{8\ F\ D_{cm}}{\Pi\ d_w^2}. \tag{6}$$

Substitution of the force from Equation (3) into Equation (6) gives the stress associated with compression to the solid height.

The force $F_y$ required to extend the spring to its yield limit, from Equation (6), is

$$F_y = \frac{\Pi\ d_w^2\ S_{ty}}{8D_{cm}K_{sc}}\ . \tag{7}$$

The spring constant may be determined with Equations (2) through (7). During the dynamic simulation, the spring stress and position are monitored, assuring that the yield stress is not exceeded in extension and the solid length is not reached in compression. Damping characteristics are usually neglected since the damping ratio of steel is approximately 0.005.

As an example, consider two parallel springs of a material with a torsional modulus of 1.1E+7 psi, an ultimate strength of 200,000 psi, with eight coils, a wire diameter of 1/4 in., an outside coil diameter of 3 in., and a free length of 10 in. Equations (2), (3), (4), and (6) yield the values in

Table 1 for the spring constant, solid length, force to compress to solid, and material stress in the solid position. Also shown in Table 1 is the stress-strain characteristics over a range of spring strain values. Note that the stress-strain characteristics are linear.

## SINGLE-ACTING AIR SPRINGS

In FY89, CSS conceived the variable deadrise hull (VDH) to increase the speed of planing boats at sea by enabling the operators to optimize the shape of the hull according to the encountered sea state. The hull in contact with the water was separated from the inner hull by an air-filled bladder that could be pressurized by the operators to achieve the desired deadrise angle. During the course of testing, the rough-water ride qualities and impact accelerations of the VDH were observed to be dramatically improved over those of the fixed hull baseline boat. The VDH is essentially a single-acting air-spring, if the structural contribution of the bag may be neglected. One of the objectives of the present CSS research is to simulate the water-impact dynamics of the VDH hull.

Examples of commercially available, single-acting air springs are the Airstroke and Airmount Isolators, manufactured by the Firestone Industrial Products Company. Other examples are the Air Cylinders manufactured by Bimba, which become single-acting when operating with one end vented to the atmosphere.

The nonlinear stiffness of an air bladder (or single-acting air spring) may be modeled by simulating the adiabatic or isothermal compression/expansion of air.[4] Adiabatic compression and expansion involves no heat transfer and is assumed for fast cycles. Isothermal compression and expansion involves constant temperature and is assumed for slow cycles. For two different pressure/volume conditions of an air spring, or two conditions of an air bladder (neglecting the structural forces from the air bladder material), the following laws apply.

$$PV = C \qquad \textit{for isothermal compression/expansion, and} \tag{8}$$

$$PV^{K} = C \qquad \textit{for adiabatic compression/expansion,} \tag{9}$$

where $k$ is the ratio of specific heats, 1.4 for air.

The nonlinear stress-strain properties of a pre-loaded, single-acting air spring (or air bladder) may be estimated in the following manner. Figure 3 shows a cylinder and piston in three positions. The device is shown in the initial unloaded position 0, pre-loaded static equilibrium position 1, and compressed position 2. The cylinder (or bladder) has a cross-sectional area $A$, initial cylinder length (or air bladder thickness) $T_o$, absolute (atmospheric) pressure $P_a$ above the piston, and absolute pressure $P_a + P_g$ within the enclosure.

Given the applied equilibrium force $F_1$, the gage pressure in the equilibrium position 1 is equal to the force $F_1$ divided by the area $A$. Applying Equation (8) to the slow isothermal application of the equilibrium force $F_1$ yields the following expression for the thickness in position 1.

$$T_1 = \frac{P_a T_o}{P_a + F_1 / A} \tag{10}$$

Next, Equation (9) is applied to the adiabatic compression of the gas from condition 1 to condition 2. Assuming the piston is displaced a distance $\delta$,

$$(P_a + P_{g1})\,(A\,T_1)^k = (P_a + P_{g2})\,(A\,T_1 - A\,\delta)^k \tag{11}$$

Solving for $P_{g2}$, and noting that the net stress $S_2$ on the piston is equal to $P_{g2}$, yields the following expression for the stress $S_2$ as a function of strain $\epsilon$ for the compression and expansion of the single-acting air spring about a pre-loaded equilibrium condition

$$S_2 = \frac{(P_a + P_{g1})\,(A\,T_1)^k}{(A\,T_1 - A\,\epsilon T_o)^k} \tag{12}$$

where the strain is based on the original unloaded thickness; i.e., $\epsilon$ is $\delta/T_o$.

Table 2 and Figure 4 show the predicted stress vs. strain characteristics of an air bladder with a cross-sectional area of 32 sq ft, an unloaded thickness of 1.0 ft (assumed to be at zero gage pressure), and an equilibrium load of 1000 lb. The equilibrium thickness under load from Equation (10) is 0.985 ft.

## DOUBLE-ACTING AIR SPRINGS

Double-acting air springs are available from several companies. An example is the series manufactured by Bimba, which they call air cylinders. These devices include a rod and sealed piston within a hollow stainless steel tube. Each end of the piston has a fitting to pressurize the volume with a source of compressed gas. If each end of the cylinder is pre-loaded to the same pressure, for example, and the piston is externally unloaded, the piston will assume a position halfway along the length of the cylinder. Positioning the load in equilibrium and changing the stiffening characteristics may be achieved by pressurizing each end with specified and unequal pressures.

The nonlinear stress-strain properties of a pre-loaded, double-acting air spring may be estimated in the following manner. Figure 5 shows a cylinder and piston in three positions: the unloaded position 0, the pre-loaded static equilibrium position 1, and the compressed position 2. The cylinder has a cross-sectional area $A$, initial cylinder length $L$, and atmospheric pressure $P_a$ above the piston. The upper and lower initial lengths are $L_{Uo}$ and $L_{Lo}$. Given the unloaded gage pressure $P_{go}$ (which must be equal for the upper and lower chambers), the initial piston position, and the pre-load force in position 1, the equilibrium position 1 may be estimated. Applying Equation (8) to the upper and lower chambers between the initial and equilibrium positions gives

$$(P_a + P_{go})\, L_{Uo}A = (P_a + P_{gU1})\, L_{U1}A \tag{13}$$

and

$$(P_a + P_{go}) L_{Lo}A = (P_a + P_{gL1}) L_{L1}A\,. \tag{14}$$

The equation relating the upper and lower pressures to the applied force is

$$F_1 = (P_{gL1} - P_{gU1})A\,. \tag{15}$$

Next, Equation (9) is applied to the process of compression beyond equilibrium, between positions 1 and 2, a displacement δ. For the upper chamber,

$$(P_a + P_{gU1})\,(L_{U1}A)^k = (P_a + P_{gU2})\,(L_{U1}A + \delta A)^k \tag{16}$$

or

$$P_{gU2} = \frac{(P_a + P_{g1})\,(L_{U1}A)^k}{(L_{U1}A + \epsilon LA)^k} - P_a, \tag{17}$$

where the strain $\epsilon$, based on total length, is $\delta/L$. For the lower chamber,

$$(P_a + P_{gL1})\,(L_{L1}A)^k = (P_a + P_{gL2})\,(L_{u1}A - \delta A)^k \tag{18}$$

or

$$P_{gL2} = \frac{(P_a + P_{gL1})\,(L_{L1}A)^k}{(L_{L1}A - \epsilon LA)^k} - P_a\,. \tag{19}$$

Finally, the total stress in position 2, $S_2$, is $P_{gU2}$ - $P_{gL2}$.

Table 3 and Figure 6 illustrate the stress-strain characteristics of a 3-in. Bimba air cylinder. The double-acting air spring is pressurized to 10 psig, and pre-loaded with 200 lb.

## SUMMARY AND CONCLUSIONS

The identification and evaluation of shock mitigation concepts for high speed planing boats requires consideration of both linear and nonlinear isolation characteristics. Some of the materials and concepts under consideration include helical springs, foams, air springs, and air bladders. Simple methods may be used for predicting the damping and nonlinear stiffening characteristics of these materials and devices based on geometry and material properties, and for implementing the characteristics within a time-domain simulation for shock mitigation concept evaluation.

$$M_h \ddot{Z}_h = C_h(\dot{Z}_d - \dot{Z}_h) + K_h(Z_d - Z_h) - F_{hg}$$

$$M_d \ddot{Z}_d = C_h(\dot{Z}_h - \dot{Z}_d) + K_h(Z_h - Z_d) + C_i(\dot{Z}_b - \dot{Z}_d) + K_i(Z_b - Z_d) - F_{dg}$$

$$M_b \ddot{Z}_b = C_i(\dot{Z}_d - \dot{Z}_b) + K_i(Z_d - Z_b) - F_{bg} + F_w$$

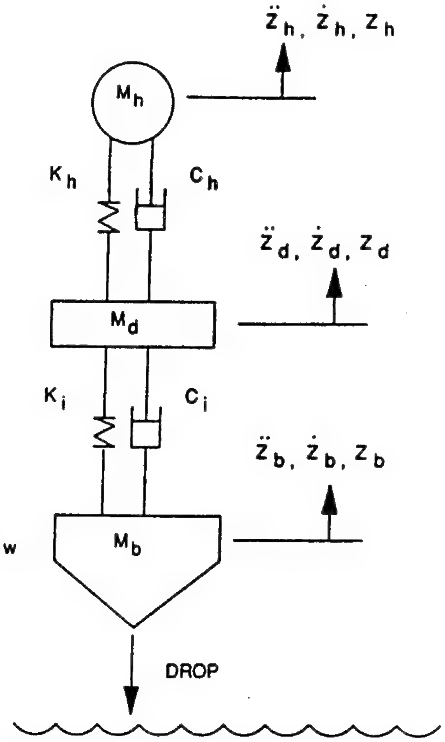


FIGURE 1. ELASTIC HULL/HUMAN DYNAMIC SYSTEM

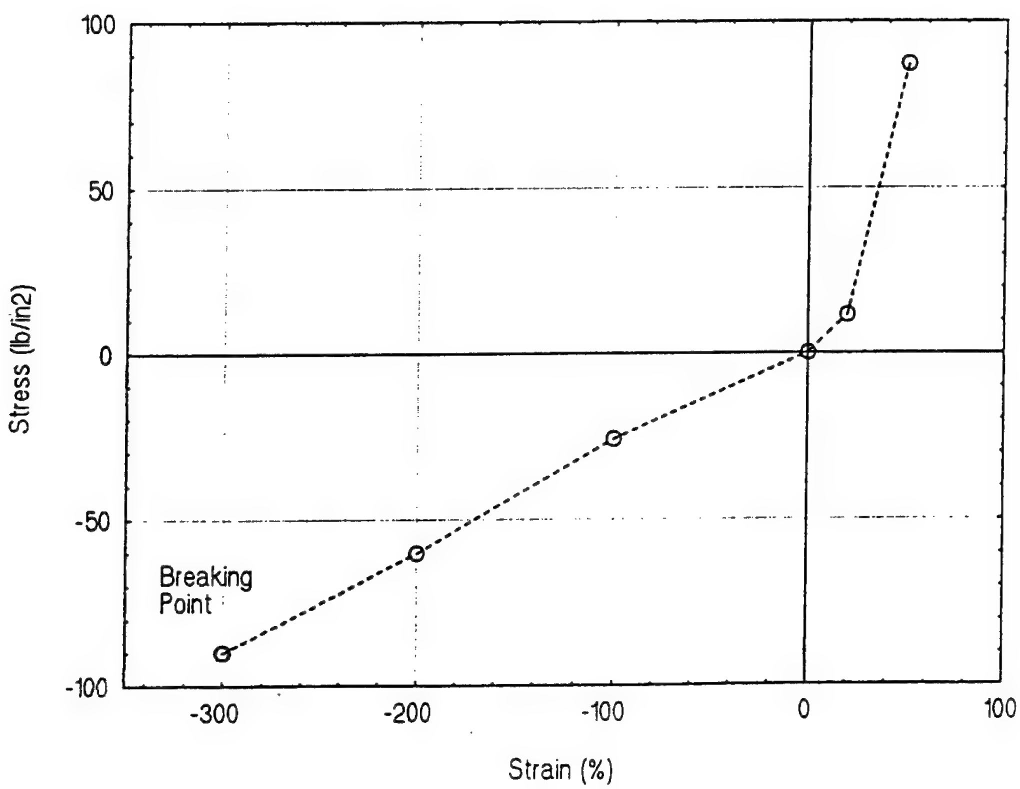


FIGURE 2. STRESS-STRAIN CHARACTERISTICS OF SORBOTHANE

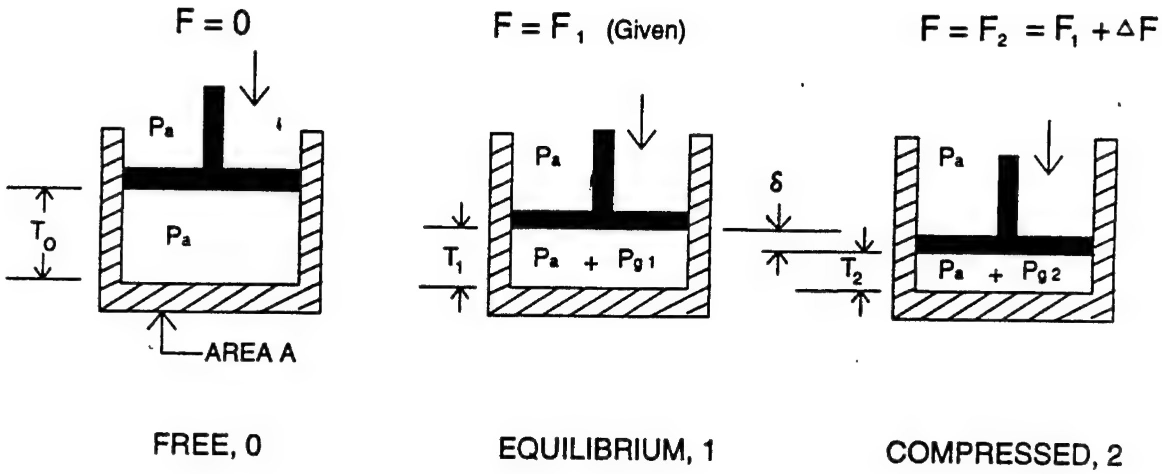


FIGURE 3. SINGLE-ACTING PRE-LOADED AIR SPRING OR AIR BLADDER

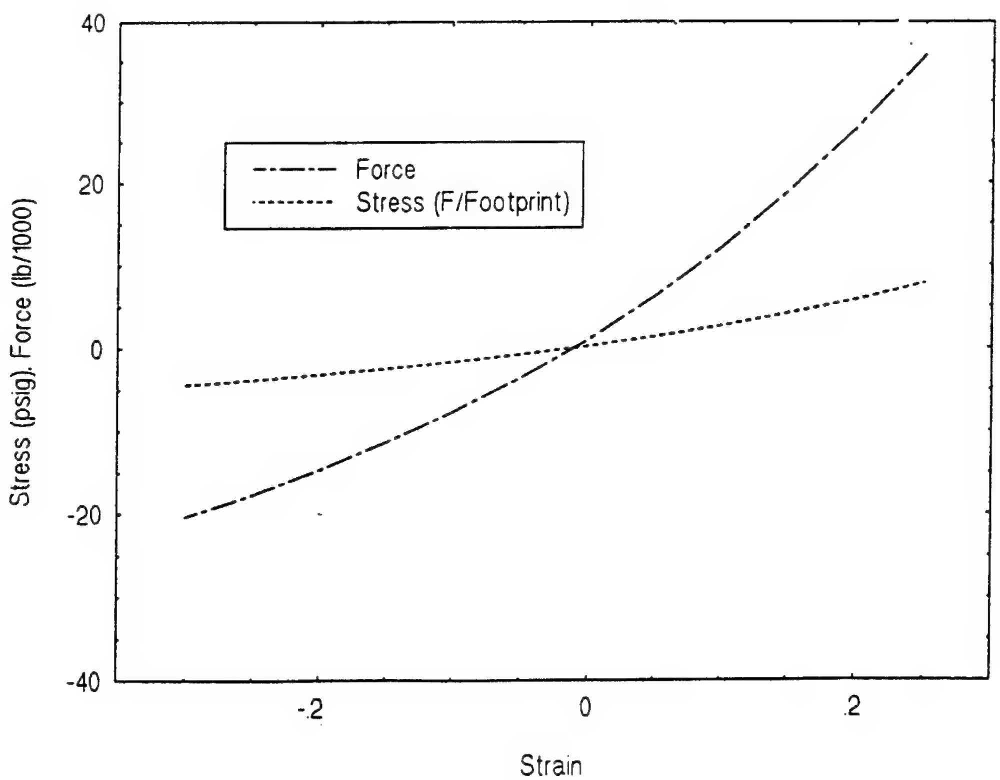


FIGURE 4. STRESS-STRAIN CHARACTERISTICS OF A 32 SQ FT AIR BLADDER

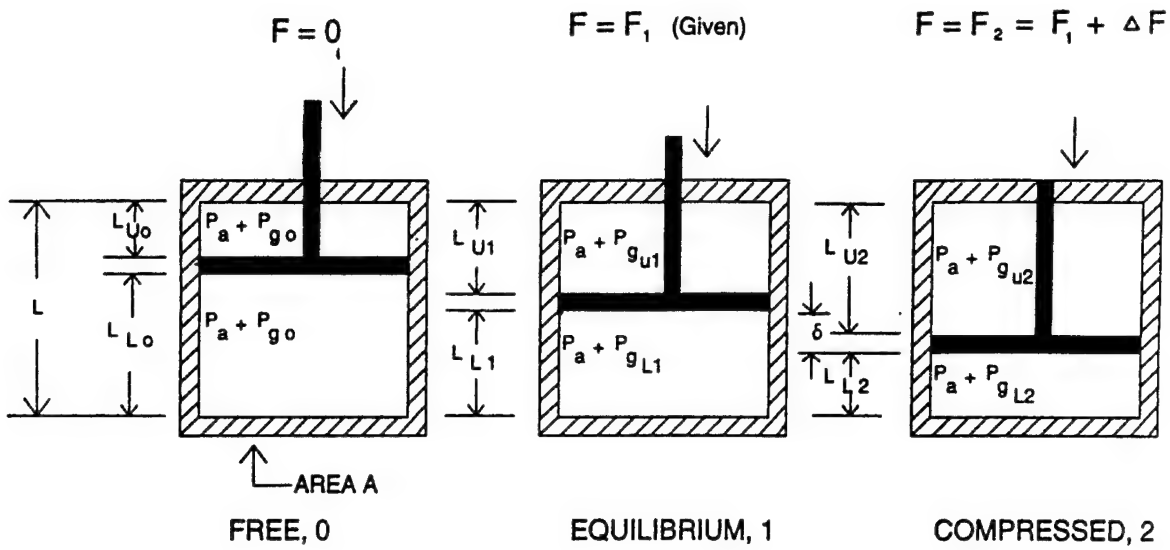


FIGURE 5. DOUBLE-ACTING AIR SPRING

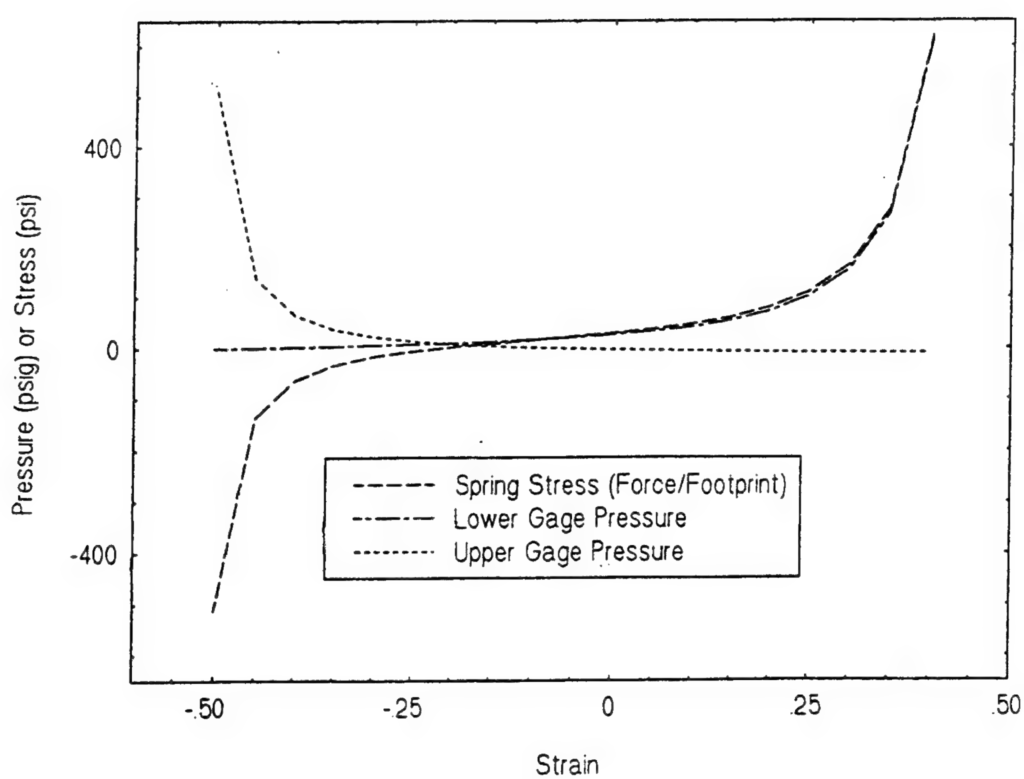


FIGURE 6. STRESS-STRAIN CHARACTERISTICS OF A 3-IN. BIMBA AIR CYLINDER

TABLE 1. DEFLECTION CHARACTERISTICS OF TWO PARALLEL HELICAL SPRINGS

**INPUTS**

| | |
|---|---|
| Torsional Modulus | 1.1E+07 psi |
| Ultimate Tensile Strength | 200000 psi |
| Wire Diameter | 0.25 in. |
| Coil Outside Diameter | 3 in. |
| No. of Coils | 8 |
| Free Length | 10 in. |
| No. of Springs | 2 |

**RESULTS**

| | |
|---|---|
| Yield Strength | 86550 psi |
| Coil Mean Diameter | 2.75 in. |
| Footprint Area, Each | 4.908735E-02 sq ft |
| All | 9.817469E-02 sq ft |
| Spring Index, Dcm/Dw | 11 |
| Shear-Stress Mult Factor | 1.130909 |
| Spring Constant per Spring | 32.28306 lb/in. |
| | 387.3967lb/ft |
| Spring Constant, All Springs | 64.56612 lb/in. |
| | 774.7934 lb/ft |
| Solid Height | 2.25 in. |
| Distance to Compress Solid | 7.75 in. |
| Force to Compress Each to Solid | 250.1937 lb |
| All to Solid | 500.3874 lb |
| Material Stress, Solid Position | 126811.1 psi |

Force/Deflection/Stress Characteristics (All Springs):

| Total Force (lb) | Spring Deflect (in.) | Material Stress (psi) | Spring Strain (%) | Footprint Stress (psi) |
|---|---|---|---|---|
| 0.0 | 0.0 | 0.0 | 0.0 | 0.0 |
| 50.0 | 0.8 | 12671.3 | 7.7 | 3.5 |
| 100.0 | 1.5 | 25342.6 | 15.5 | 7.1 |
| 150.0 | 2.3 | 38013.9 | 23.2 | 10.6 |
| 200.0 | 3.1 | 50685.2 | 31.0 | 14.1 |
| 250.0 | 3.9 | 63356.4 | 38.7 | 17.7 |
| 300.0 | 4.6 | 76027.7 | 46.5 | 21.2 |
| 350.0 | 5.4 | 88699.0 | 54.2 | 24.8 |
| 400.0 | 6.2 | 101370.3 | 62.0 | 28.3 |
| 450.0 | 7.0 | 114041.6 | 69.7 | 31.8 |
| 500.0 | 7.7 | 126712.9 | 77.4 | 35.4 |
| 550.0 | 8.5 | 139384.2 | 85.2 | 38.9 |
| 600.0 | 9.3 | 152055.5 | 92.9 | 42.4 |

TABLE 2. STRESS-STRAIN CHARACTERISTICS OF A 32 SQ FT AIR BLADDER

AIR SPRING OR BLADDER STRESS-STRAIN CHARACTERISTICS
SINGLE-ACTING, WITH PRE-LOAD

| | |
|---|---|
| Unloaded Gage Pressure | 0 psi |
| Spring/Bladder Area | 32 sq ft |
| Thickness | 1 ft |
| Volume | 32 cu ft |
| Equilibrium Load | 1000 lb |
| Gage Pressure | 31.25 lb/sq ft |
| Thickness | .9854519 ft |
| Volume | 31.53446 cu ft |

Stress-Strain Characteristics About Equilibrium
(Strain based on Free Thickness)

| Strain | Deflection ft | Isothermal Stress psig | Isothermal Force lb | Adiabatic Stress psig | Adiabatic Force lb |
|---|---|---|---|---|---|
| -0.40 | -0.40 | -4.09 | -18845.54 | -5.44 | -25074.15 |
| -0.35 | -0.35 | -3.69 | -17015.00 | -4.95 | -22821.25 |
| -0.30 | -0.30 | -3.26 | -15042.05 | -4.42 | -20356.43 |
| -0.25 | -0.25 | -2.80 | -12909.40 | -3.83 | -17650.28 |
| -0.20 | -0.20 | -2.30 | -10596.86 | -3.18 | -14667.92 |
| -0.15 | -0.15 | -1.75 | -8080.65 | -2.47 | -11367.64 |
| -0.10 | -0.10 | -1.16 | -5332.62 | -1.67 | -7699.19 |
| -0.05 | -0.05 | -0.50 | -2319.21 | -0.78 | -3601.57 |
| 0.00 | 0.00 | 0.22 | 1000.00 | 0.22 | 1000.00 |
| 0.05 | 0.05 | 1.01 | 4674.03 | 1.35 | 6198.06 |
| 0.10 | 0.10 | 1.90 | 8763.00 | 2.63 | 12108.37 |
| 0.15 | 0.15 | 2.90 | 13341.39 | 4.10 | 18877.56 |
| 0.20 | 0.20 | 4.02 | 18502.69 | 5.79 | 26693.82 |
| 0.25 | 0.25 | 5.29 | 24365.77 | 7.77 | 35802.36 |
| 0.30 | 0.30 | 6.75 | 31084.21 | 10.10 | 46528.18 |
| 0.35 | 0.35 | 8.43 | 38859.93 | 12.87 | 59310.52 |

TABLE 3. STRESS-STRAIN CHARACTERISTICS OF 3-IN. BIMBA AIR CYLINDER

DOUBLE-ACTING AIR SPRING CHARACTERIZATION WITH PRE-LOADING EFFECT INCLUDED

| | |
|---|---|
| **INPUTS** | |
| Spring Geometry | |
| Diameter | in.3 |
| Area | 4.908735E-02 sq ft |
| O/A Length | 1.666667 ft |
| Unloaded | |
| Free Upper Length | .4166667 ft |
| Free Lower Length | 1.25 ft |
| Pressure | 10 psig |
| Load | 200 lb |
| **RESULTS** | |
| Equilibrium Upper Length | .891629 ft |
| Lower Length | .7750376 ft |
| Upper Gage Pressure | -454.6739 psig |
| Lower Gage Pressure | 3619.696 psig |
| Spring Constant | 589.5203 lb/ft |

| Strain O/A ft/ft | Deflection ft | Pressure Upper psig | Pressure Lower psig | Stress psig | Force lb |
|---|---|---|---|---|---|
| -0.50 | -0.83 | 510.92 | -0.37 | -511.28 | -3614.03 |
| -0.45 | -0.75 | 136.99 | 0.74 | -136.24 | -963.04 |
| -0.40 | -0.67 | 64.66 | 2.01 | -62.65 | -442.87 |
| -0.35 | -0.58 | 36.35 | 3.46 | -32.89 | -232.49 |
| -0.30 | -0.50 | 21.82 | 5.14 | -16.68 | -117.89 |
| -0.25 | -0.42 | 13.18 | 7.11 | -6.06 | -42.86 |
| -0.20 | -0.33 | 7.53 | 9.44 | 1.91 | 13.51 |
| -0.15 | -0.25 | 3.60 | 12.23 | 8.64 | 61.06 |
| -0.10 | -0.17 | 0.72 | 15.63 | 14.91 | 105.38 |
| -0.05 | -0.08 | -1.46 | 19.83 | 21.29 | 150.47 |
| 0.00 | 0.00 | -3.16 | 25.14 | 28.29 | 200.00 |
| 0.05 | 0.08 | -4.51 | 32.01 | 36.53 | 258.21 |
| 0.10 | 0.17 | -5.62 | 41.21 | 46.83 | 331.03 |
| 0.15 | 0.25 | -6.53 | 54.02 | 60.55 | 428.01 |
| 0.20 | 0.33 | -7.30 | 72.83 | 80.13 | 566.40 |
| 0.25 | 0.42 | -7.95 | 102.59 | 110.54 | 781.39 |
| 0.30 | 0.50 | -8.51 | 155.20 | 163.71 | 1157.19 |
| 0.35 | 0.58 | -8.99 | 266.91 | 275.91 | 1950.27 |
| 0.40 | 0.67 | -9.42 | 611.13 | 620.55 | 4386.41 |
| 0.45 | 0.75 | -9.79 | 4852.84 | 4862.63 | 34371.85 |

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